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TESTING OF A "DURA SEAL" ROTARY SEAL

by

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ABSTRACT

Tests were run to determine the effectiveness of air cooling, the operating characteristics, and the life expectancy for a $5\frac{1}{2}$ in. Dura Seal. It was found that air cooling was satisfactory. The power consumed by the seal was generally less than 1000 watts and the leakage of air through the seal less than 0.002 lb/sec. The rate of wear of the face of the graphite member varied considerably with conditions; however, it appeared that satisfactory performance can be attained only in an installation where the seal is readily accessible and can be adjusted periodically.

INTRODUCTION

One of the most important problems encountered in designing a pump for circulating solutions of radioactive materials is that of selecting the shaft seal. Selection of the proper seal is important from the standpoint of preventing the leakage of radioactive gas or liquid from the pump housing to uncontaminated areas. A Dura Seal rotary seal, manufactured by the Durametallic Corporation, of Kalamazoo, Michigan, was selected as the seal for a vertical pump designed to utilize gas pressure for preventing radioactive solutions from entering the seal.

The Dura Seal consisted of a stellite member fastened to a $5\frac{1}{2}$ in. diameter pump shaft and a stationary graphite member which surrounded the shaft and was sealed to the pump housing. A seal to prevent escape of gas from the housing was effected by contact between ground faces of stellite and graphite. Standard Dura Seals are cooled by circulating water through the graphite member; however, it was considered for that member to break in a way that would permit the coolant to dilute the circulating solution, so this seal was designed for air cooling.

Tests described in this report were made to determine the effectiveness of air cooling, the operating characteristics, and the life expectancy of the $5\frac{1}{2}$ in. air cooled Dura Seal.

EXPERIMENTAL

A satisfactory pump was not available for testing the Dura Seal, so apparatus was assembled, as shown in Figure 1, for obtaining the

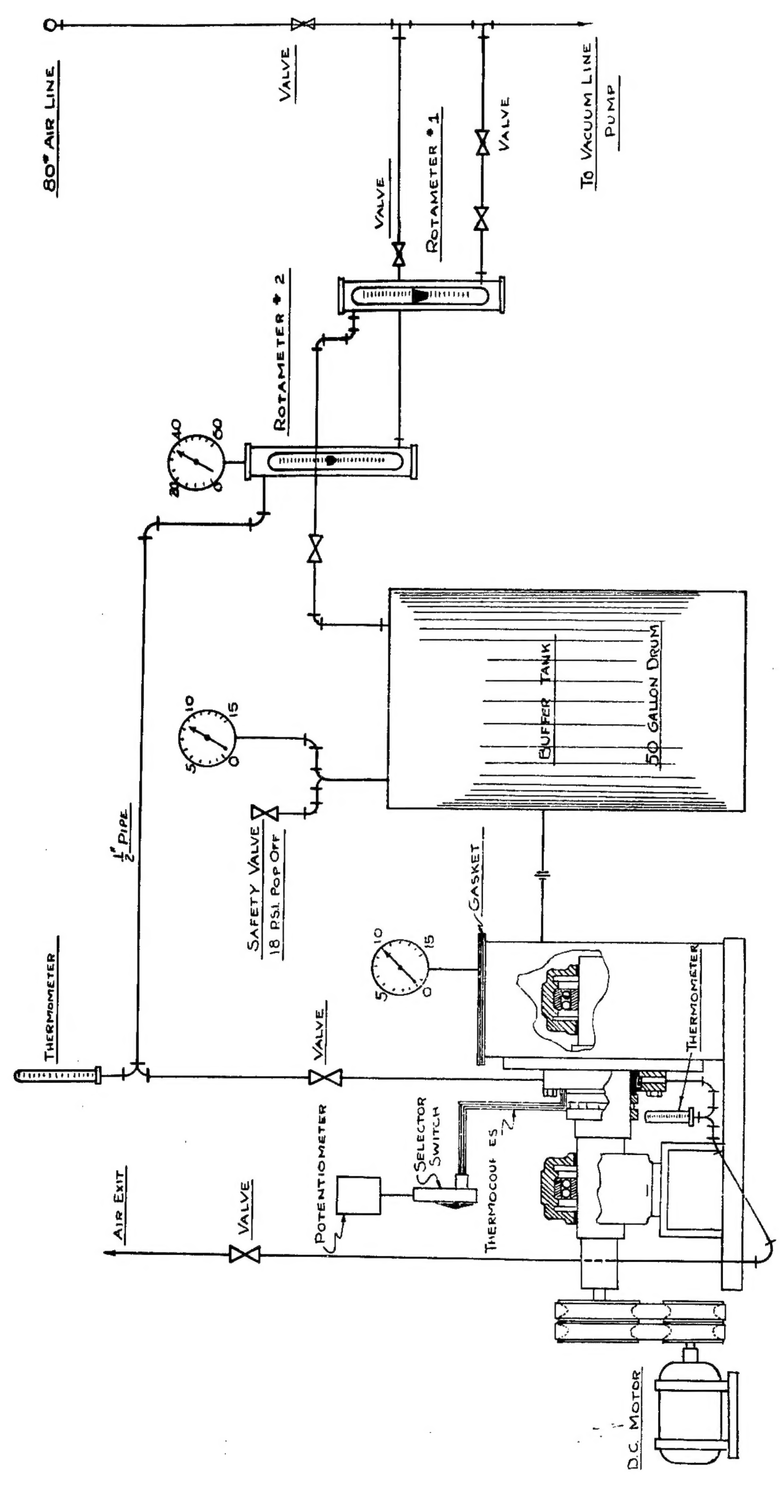

Figure 1. Diagram of test apparatus.

necessary data. The pump housing was simulated by a steel box. A $5\frac{1}{2}$ in. diameter shaft was mounted between bearings, one in the box and one outside of the box; the seal was mounted on the shaft and on the box at the opening. Air and vacuum lines were connected to the box through a 50 gal. drum used as a buffer to smooth the pressure fluctuations, and air was piped to cool the graphite member of the seal. The shaft was driven through belts by a DC motor to permit easy adjustment of speed and determination of power consumed in the seal. Rotameters and pressure gauges were installed in the lines for measuring the flow of cooling air and the seal leakage. Thermometers were installed in the air lines, and thermocouples were installed in the graphite insert for measuring the temperature of the seal. Figure 2 is a drawing of the seal showing the location of the thermocouples.

Prior to making a run, the rotating member of the seal was adjusted to a predetermined compression of the springs holding the seal members in contact. The motor was then run at a specific speed or power output. Seal temperatures and leakage and cooling air temperatures and flow rate were recorded for given pressures in the box.

At the conclusion of the initial performance tests the seal was removed, and the springs were tested to determine the relationship between the spring setting and the force between the seal members. The unit was reassembled, and long term tests were run, during which time the rate of wear of the seal was determined as a function of spring setting and box pressure for a constant speed of 700 rpm.

RESULTS

The initial object of the tests was to determine the adequacy of air cooling for the Dura Seal. Plotted in Figure 3 are curves for the maximum temperature of the graphite face as a function of the flow of cooling air and the shaft speed. The data were taken during tests when there was no air pressure acting against the seal and when there was no air flowing between the faces. The spring setting was maintained at 0.375 in. which corresponds to a force of 150 lbs exerted in holding the faces of the seal in contact. The relationship between the spring setting and the force exerted by the springs is shown in Figure 4. Coolant pressure data for air flowing through the coolant chamber in the graphite are plotted in Figure 5.

It was recommended by Messrs Gregory and Van Blaricom, of the Durametallic Corporation, that the temperature of the graphite member of the seal be kept below 225 °F. Experience had shown that trouble could be expected with the neoprene packing in the rotary member of the seal if that temperature was exceeded. It is evident from the curves in Figure 3 that a flow of at least 0.070 lbs/sec of air is required at shaft speeds greater than 800 rpm if the graphite temperature is limited to 225 °F. Tests were run at higher temperatures and trouble was experienced with the neoprene packing in the rotary member sticking to the shaft.

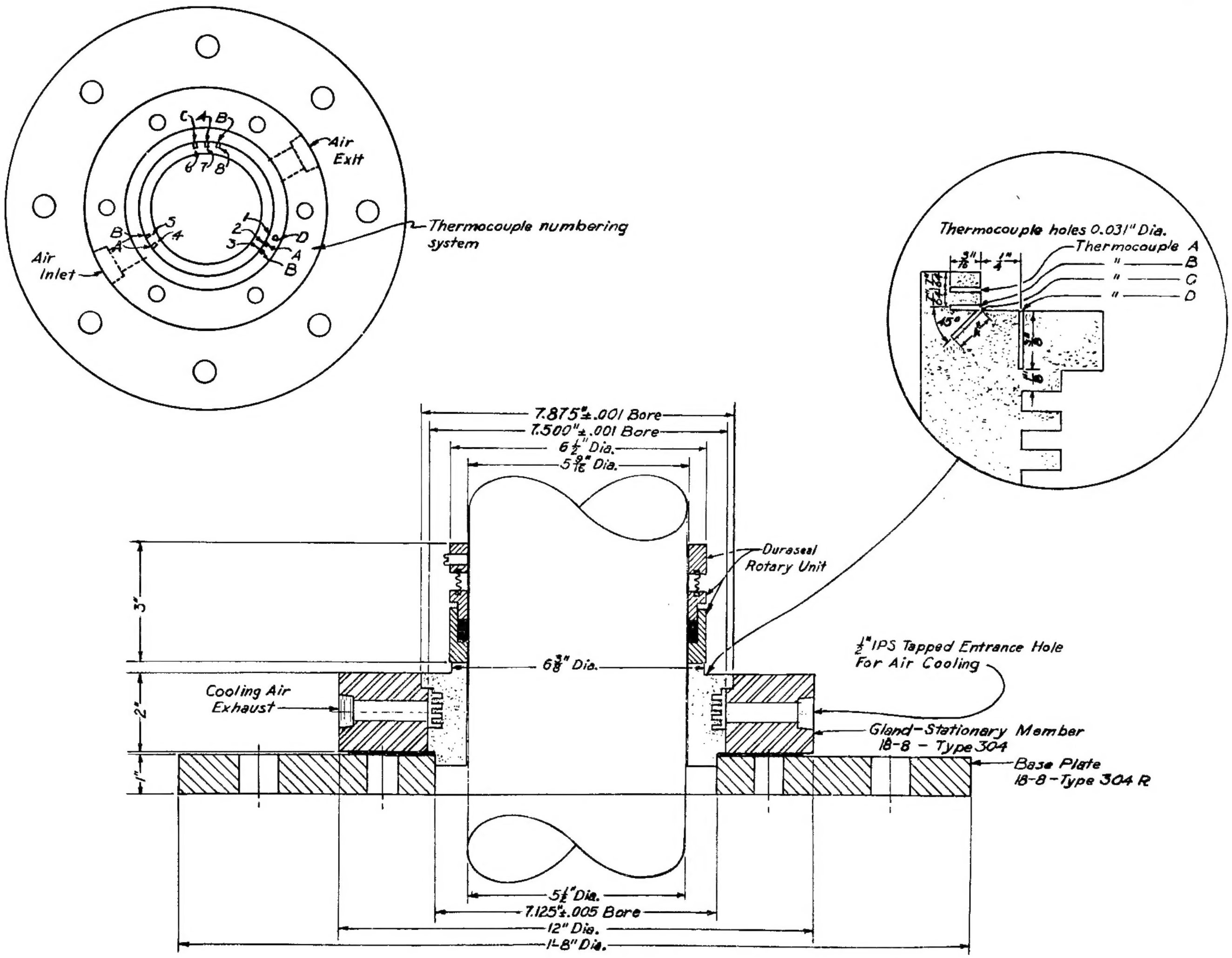

Figure 2. Thermocouple locations modified air cooled gland insert assembly.

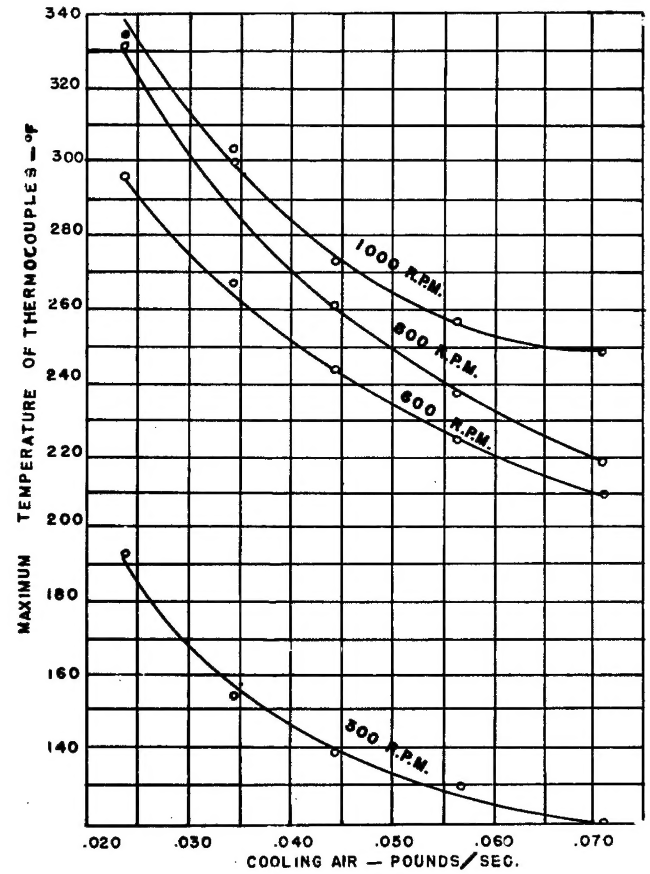

Figure 3. Maximum temperature of Dura Seal as a function of flow of coolant and shaft speed. Box pressure - 0 lb/in.^{2}, spring setting .375 in.

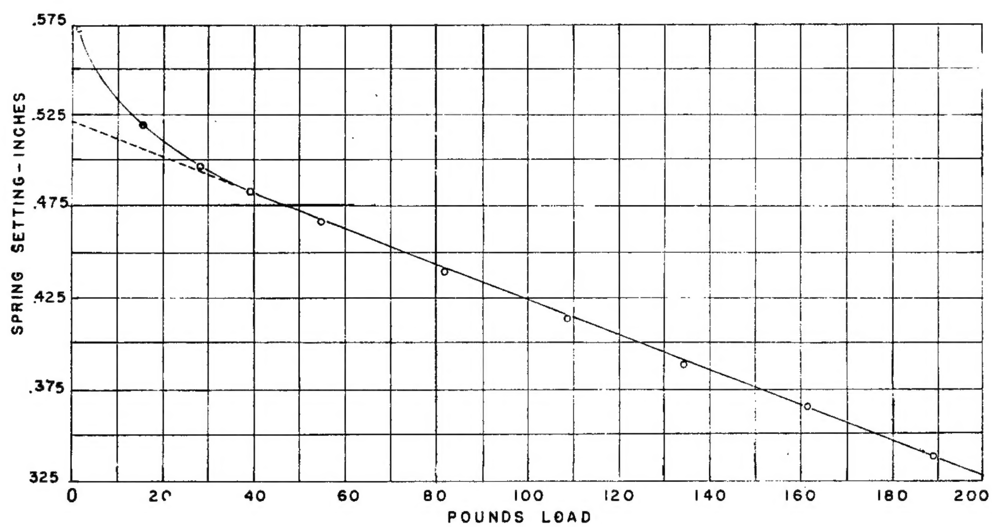

Figure 4. Durametallic Dura Seal. Spring setting vs pounds load, 20 springs total.

As a result the rotary member could not slide and adjust itself to maintain contact between the stellite and graphite faces. Graphited asbestos and then Telflon, supplied by the Durametallic Corporation, were substituted for the neoprene, and the Dura Seal was operated without difficulty at temperatures as high as 330°F.

Since the temperature of the graphite is dependent upon the power consumed by the seal for any given coolant flow, the consumption of power was measured as a function of shaft speed and spring setting. The data plotted in Figure 6 show the power consumed by the seal to be small, generally below 1000 watts. A decrease in the power consumption is indicated as the speed is increased from 1000 to 1200 rpm. It is thought that this rather unusual occurrence may be caused by the vibrations observed as the speed is increased or by a change in the coefficient of friction which, for dry surfaces, has been found to decrease with increasing speed.

Changing the air pressure in the box from atmospheric to greater or less than atmospheric pressure had a marked effect on the power consumed by the seal. For a specific spring setting and shaft speed, increasing the pressure in the box decreased the power consumed; reducing the pressure below one atmosphere had the opposite effect. Data for the relation between box pressure and power consumption for constant spring settings and shaft speeds are presented in Table 1. It appears that pressure in the box acts against the force applied by the springs and that reduced pressures increase the force holding the gland faces together. The effective area over which the pressure acts is calculated to be 8 to 10 in^2.

Data were taken during one series of tests to show the effect of box pressure and shaft speed on the quantity of air leaking through the seal. They are presented in Table 2. The variation in leakage, in lbs of air/sec, with shaft speed for a given pressure was more marked than the variation in leakage with pressure for a given shaft speed. This could be attributed to vibration of the seal at the higher shaft speeds. Under operating conditions where the box pressure balanced or exceeded the load applied by the springs, leakage increased markedly.

For the tests reported in Table 2 the leakage varied between 0.0005 and 0.0020 lbs/sec, 0.4 to 1.6 ft^3/min, at 68°F and 1 atmosphere. It was observed that the leakage varied considerably from day to day unless the seal was operated continuously, which may account for the seemingly incongruous results obtained for leakage under vacuum. That much of the air leaked through the packing that sealed the rotary member of the seal to the shaft was indicated by a decrease in the leakage when the change was made from graphited asbestos to Teflon packing.

The last tests run with the Dura Seal were those to determine the life expectancy. Date for those tests are given in Table 3. In running the tests the spring setting was adjusted to 0.375 in. at the start of each run, and no adjustments were made until the setting reached 0.415 in. It was found that the graphite face wore at rates between 0.01 and 0.19 in./month. The stellite face wore approximately 0.003 in. during the entire testing program.

Table 1. Effect of box pressure and shaft speed on power consumed by Dura Seal. Spring setting-0.375 in.

Gauge Pressure, lbs/in^2	-14.2	0	3	6	9	12	15
Watts at 300 rpm	750	435	358	233	214	-	-
Watts at 600 rpm	935	740	645	425	225	*700 rpm 129	105
Watts at 800 rpm	1000	880	630	655	355	170	113
Watts at 1000 rpm	1030	1060	790	860	630	335	152
Watts at 1200 rpm	1010	970	730	810	690	610	275

Table 2. Effect of box pressure and shaft speed on air leakage through Dura Seal. Spring setting-0.375 in.

Gauge Pressure, lbs/in^2	-14.2	3	6	9	12	15
Leakage* at 300 rpm	0.0013	0.0006	0.0007	0.0006		
Leakage at 600 rpm	0.0013	0.0009	0.0008	0.0006	0.0005	0.0007
Leakage at 800 rpm	0.0013	0.0016	0.0019	0.0007	0.0005	0.0007
Leakage at 1000 rpm	0.0014	0.0016	0.0017	0.0020	0.0006	0.0007
Leakage at 1200 rpm	0.0013	0.0015	0.0018	0.0019	0.0019	0.0020

*lbs/sec at 68°F.

Table 3. Data for Life of Dura Seal Tests. Shaft speed=700 to 800 rpm, initial spring setting=0.375 in. final spring setting=0.415 in. max.

Run No.	Duration, hrs	Box air pressure, lb/in^2	Air leakage, lbs/sec at 77°F and 14.7 lb/in^2 Max.	Ave.	Flow of cooling air, lbs/sec	Max. temp., °F	Graphite Face Wear in.	in./mo
1	168	0	0.0	0.0	0.0445	266	0.015	0.065
2	244	0	0.0	0.0	0.0445	210	0.043	0.127
3	135	5	0.00015	0.0001	0.0445	175	0.036	0.192
4	146	15	0.0012	0.0003	0.0445	105	0.002	0.010
5	420	12	0.0003	0.00016	0.0445	160	0.045	0.076

The rate of wear varied considerably, but air leaking between the faces of the seal to provide lubrication and some cooling did not appear to be an important factor. The only satisfactorily low rate of wear occurred when the pressure in the box was maintained at 15 lbs/in^2, to almost balance the force exerted by the springs. Although the wear tests were run at a shaft speed of 700 to 800 rpm, it is believed that the wear is proportional to the power consumed by the seal at speeds below 1000 rpm and that the rate of wear at shaft speeds less than 800 rpm can be predicted on that basis.

CONCLUSIONS

It can be concluded from the tests that for the $5\frac{1}{2}$ inch Dura Seal:

1). Air cooling is adequate. Less than 0.07 lb/sec of air will be needed and the pressure drop in the seal will be less than 15 lbs/in^2.

2). For the application under consideration, the power consumption will be less than 1000 watts.

3). Leakage of air between the faces of the seal can be kept at less than 0.002 lb/sec and probably at less than 0.0002 lb/sec if necessary.

4). Teflon should be used as the shaft packing for the rotating member.

5). For good performance, the shaft speed should be less than 1000 rpm.

6). Adjustment of the spring settings will probably be the only maintenance required by the seal.

The life of a Dura Seal is difficult to estimate. Depending upon the installation and operating conditions, it may be a few weeks with adjustments of the springs required every few days, or it may be one or more years with spring adjustment required only at intervals of several months. On the basis of the present knowledge regarding the intended applications, it seems probable that the life will not be greater than six months and that it will be necessary to adjust the seal at least once each month. If such a seal is used, it appears necessary that it be readily accessible, so that the adjustments can be made with a minimum of difficulty.

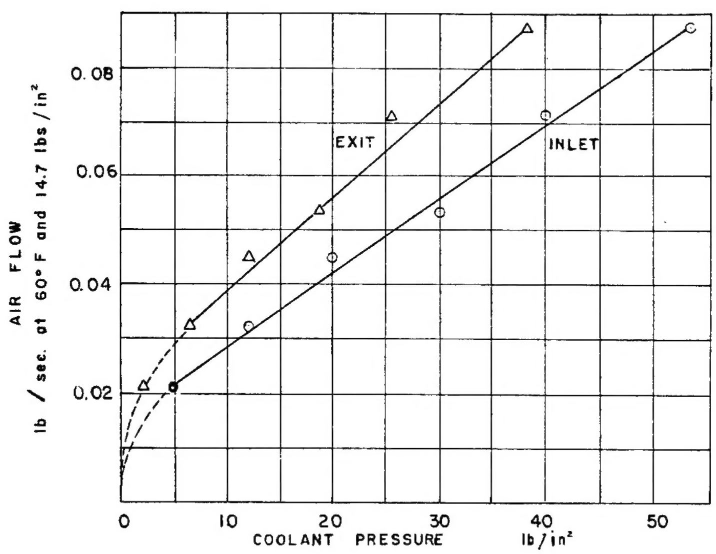

Figure 5. Coolant pressure data for Dura Seal.

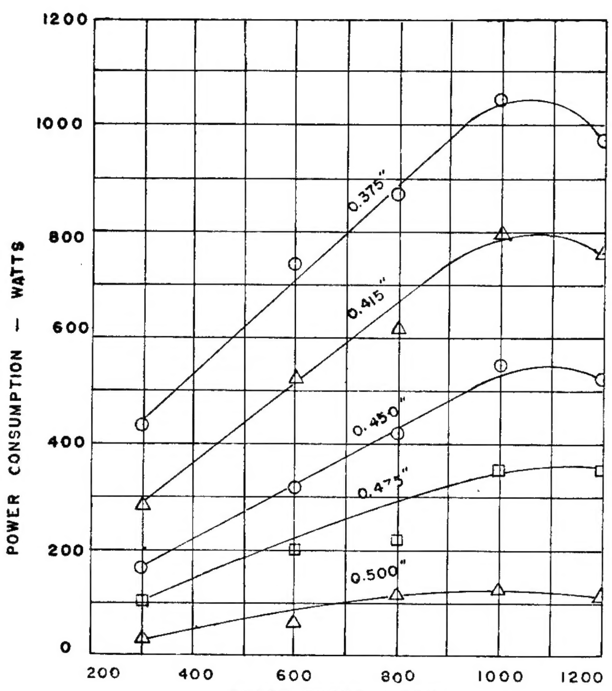

Figure 6. Power consumed by Dura Seal as a function of shaft speed and spring setting. Box pressure, 0 lb/in.^{2}.

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